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Performance Modeling of a Solar Driven Absorption Cooling System for Carnegie Mellon University's Intelligent Workplace

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Abstract:The Robert L. Preger Intelligent Workplace (IW) is a 650 m^2 (7,000 ft^2) living laboratory of office space at Carnegie Mellon University (Pittsburgh, PA). The IW is involved in a project to develop, install, and test an effective solar thermal system for space heating and cooling. The proposed energy supply system configuration includes integrated compound parabolic concentrator (ICPC), a hot storage tank, a gas fired auxiliary heater, a steam generator, a steam driven absorption chiller and fan coils. A TRNSYS predictive model has been programmed and used to evaluate the performance of the system throughout a summer season. The effects on performance and on costs have been explored for various design variables and operating conditions.

The performance calculations indicate that:

- the 16.17 kW (55.2 kBtu/hr, 4.5 tons) absorption chiller is adequate to meet the IW south cooling requirements
- 30-40m^2 collectors can supply from 55 to 65% of the heat required to drive the chiller
- estimated heat losses from the system can reach about 20-30% of the total heat collected.

1. INTRODUCTION

The world demand of energy for air conditioning is continually increasing. As traditional cooling units are electrically powered, peak demand of electrical power in summer increases and tends to reach the capacity limit in some countries. Because most of the electrical power stems from fossil fired power plants this also augments the production of CO_2. A more innovative approach to provide cooling is to use solar energy in a heat driven active absorption cycle for air conditioning. The high correlation between the availability of solar energy and the need for cooling in a building provides an inherent advantage to solar driven cooling. In addition, absorption based systems have the advantage of the using reject heat from power generation as well as solar heat. The cooling supply system for building through absorption refrigeration makes direct and efficient use of solar heat, replacing the use of natural gas or electrical energy for compression refrigeration. Approximately 50 to 60 % of the radiant energy impinging on the receivers is passed to the heat transfer medium for use. The coefficient of performance of absorption chiller, the ratio of heat removed in cooling to the heat supplied to the system is 1.1 to 1.2 for a two stage chiller. The performance of a solar driven absorption cooling system was studied and modeled in TRNSYS for an office space in Pittsburgh (USA) named "the Intelligent Workplace". The location of the IW is not an ideal place for solar application due to its latitude and frequently cloudy weather. Moreover the small scale of the solar driven cooling system militates against an economic application of the technology. Nonetheless, the TRNSYS model and the simulation evaluation can explore how solar energy through high temperature solar receivers and heat storage might be integrated into efficient energy supply system for a building, even in a place like Pittsburgh (latitude 40.3°N, longitude 79.6°W).

1.1 Building description

The IW is a 650 m^2 (7,000 ft^2) living laboratory of office space at Carnegie Mellon University. The IW south zone's net floor area is about 245 m^2 (2,637 ft^2). The average height is about 4.8 m (15.8 ft), including the raised floor and average height of roof. The open space is subdivided by partition walls and furniture in 10 office or conference spaces on the south, east and west parts. The building has horizontal shadings on the east and west facades. The IW_s floor plan is shown in Figure 1.

1.2 Internal Loads

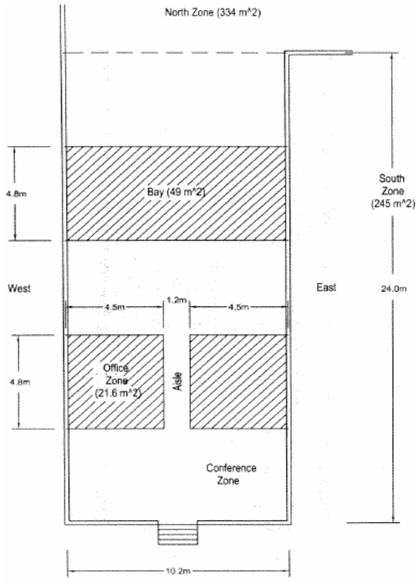


Fig. 1. IW south Floor Plan

The IW_s is occupied all week long by faculty and students. The internal loads in the IW_s including lighting, plug loads such as computers, and occupants contribute to the overall cooling requirements. The maximum occupancy is 30 people. Most of the occupant arrived between 9AM and 10 AM and left the building between 5 PM and 8PM. The total heat gain of the artificial lighting is 17 W/m^2 (5.4 Btu.hr^{-1}. ft^{-2}). The equipment heat gain is 100 W (341 Btu/hr) per computer (one per person) and 50 W (170 Btu/hr) for one printer. The equipment and lighting heat gain are defined based on the occupancy schedule.

1.3 HVAC System

Ventilation and air conditioning are split to reduce the amount of air treated: ventilation handles latent loads whereas air conditioning handles sensible loads.

1.3.1 Ventilation.

Fresh air is supplied to the IW_s by a desiccant wheel unit (SEMCO unit) via diffusers in the floor. The outside air volume rate is about 34m^3/hr (20 cfm) per person based on the ASHRAE standards. The outside air is supplied during the whole year with -3°C to -6°C under the set room temperature with a proper relative humidity to maintain the room conditions in the comfort range In summer, the conditioned outside air temperature is 16°C (61°F) and relative humidity is around 30 %.

1.3.2Air Conditioning.

A steam driven two stage absorption chiller has been installed in the IW and tested. Design and modeling are performed before the procurement of additional devices (solar collectors and storage tank). In the proposed system, solar radiation will be captured by high temperature solar receivers mounted on the IW roof. This thermal energy will be transferred to heat transfer medium. This fluid medium will flow to a steam generator through an auxiliary natural gas fired heater or alternatively to a hot fluid storage tank. The steam will be utilized in a two stage 16.17 kW (55.2 kBtu/hr) absorption chiller to provide chilled water to the cooling coils. The IW_s solar driven cooling system is shown in Figure 2.

2. MODEL APPROACH

The software program selected to model the IWs and its solar driven cooling system is TRNSYS. This transient systems simulation program is developed by the Solar Energy Laboratory at the University of Wisconsin. This software supports detailed simulations of multi zone buildings and their energy supply equipment. The TRNSYS equipment library includes: sophisticated building models; many of the equipment components commonly found in building thermal energy systems; components that facilitate input of weather data and of occupancy, equipment, and set point schedules; and components that support output of simulation results. Exercise of the TRNSYS simulation tool enables the search for an optimal system configuration, equipment design and operational mode. To simulate solar driven cooling system, two TRNSYS simulations are performed: the building and the energy supply system simulations.

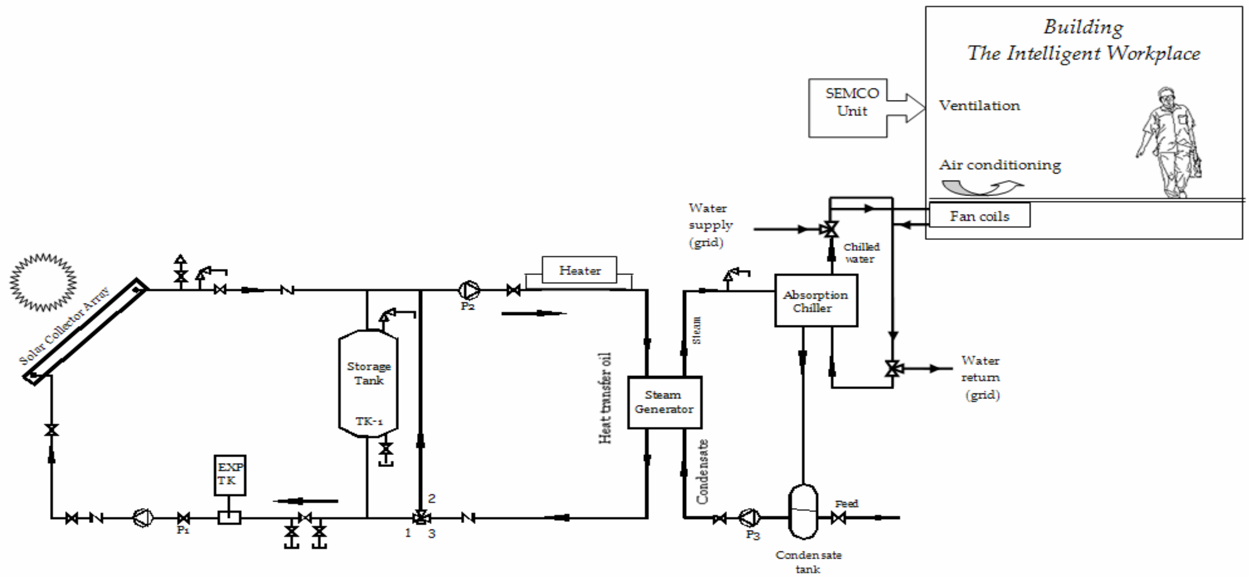


Fig. 2. Proposed IW solar driven absorption system

2.1 Building simulation

The purposes of the building simulation are to:

-simulate the IW_s zone: its geometry and materials for windows, walls, and floors; shading; ventilation supply from the ventilation unit.

-calculate the IW_s cooling and heating loads based on the supply of conditioned fresh air from the ventilation unit; the IW operating schedules for occupancy, lighting, equipment, and set points; and the Pittsburgh weather. The calculated loads are shown in Table 1.

Tab. 1. Heating and cooling loads of the IW_s

Heating season	Energy consumption	Peak load
	9,056 kWh (30.9 MBtu)	20.8 kW (71 kBtu/hr)
Cooling season	Energy consumption	Peak load
	15,176 kWh (51.8 MBtu)	14.1 kW (48 kBtu/hr)

The cooling peak load (air conditioning) is 14.1kW (48 kBtu/hr) which means 56 W/m^2 (17.7 Btu.hr^{-1}.ft^{-2}) for the air conditioning (sensible loads). The 16.17 kW (55.2 kBtu/hr) absorption chiller is then adequate to meet the IW south cooling requirements. The total cooling peak load (air conditioning and ventilation) is 25kW (85.3 kBtu/hr) which means 100 W/m^2 (31.7 Btu.hr^{-1}.ft^{-2}) for the air conditioning and ventilation (sensible and latent loads).

2.2 Energy supply system simulation

The purposes of the energy system simulation are to:

- simulate the solar, steam and load loops of the solar thermal system

- validate the design assumptions concerning the adequacy of the solar receiver area, the tank capacity, and the chiller capacity to meet the IW_s cooling loads

- validate the system operating rules and feed forward control to adjust properly the system depending on the weather conditions, the solar energy stored in the tank and the building loads

- define a base case and calculate the solar, natural gas, and electrical energy inputs to the cooling system and any required chilled water input from the grid required to meet the loads calculated by the building simulation.

- estimate capital and operating costs of a solar driven cooling system for the IW_s

- discover ways in which the capital and operating costs might be reduced without affecting the effectiveness of the cooling system by improving the base case.

3. TRNSYS MODEL

The TRNSYS simulations are carried out for the summer season (15th April - 15th October). The simulation time step is set to 15 min. This small value is chosen based on the time step of the system controller. The tolerance integration and convergence parameters are set to 0.001. Energy balance calculations were performed to control the convergence and accuracy of the results.

3.1 Model Assumptions

3.1.1 Weather. To simulate the solar cooling system, an accurate climatic data base (TMY2 file) is used to get the solar radiation, the dry bulb temperature and the relative humidity of the outside air in Pittsburgh (latitude 40.3°N, longitude 79.6°W). The solar radiation on a horizontal surface is between 500 and 900 W/m^2 (160-285 $Btu.hr^{-1}.ft^{-2}$) during the cooling season and between 200 and 500 W/m^2 (65-160 $Btu.hr^{-1}.ft^{-2}$) during the rest of the year.

3.1.2Fluid medium. The heat transfer fluid used in the solar loop and the steam generation loop is a silicone oil called Syltherm 800. This fluid has a low vapor pressure; it can be heated at atmospheric pressure in the receiver up to 400˚C (750°F) without vaporization. Its freezing point is below -40˚C (-40°F). This low freezing temperature enables the solar system to operate during the winter in Pittsburgh. While its heat capacity is lower, viscosity higher, and cost significantly higher than water, its lower pressure requirement offers a significant operating advantage and facilitates identifying a suitable pump for its circulation through the receivers. The characteristics of this fluid at 175°C (347°F) are presented in Table 2.

Tab. 2. Syltherm 800 characteristics

Specific Heat	1.881 $kJ.kg^{-1}.K^{-1}$	*0.449 $Btu.lb^{-1}.°F^{-1}$*
Density	800 kg/m^3	*49.9 lb/ft^3*
Thermal conductivity	0.3744 $kJ.hr^{-1}.m^{-1}.K^{-1}$	*0.06 $Btu.hr^{-1}.ft^{-1}.°F^{-1}$*
Dynamic viscosity	4.662 $kg.m^{-1}.hr^{-1}$	*3.132 $lb.ft^{-1}.hr^{-1}$*
Expansion coefficient	0.00026 1/K	*0.000144 1/°F*

3..1.3Solar collectors. There are many types of solar collectors, which are used in air-conditioning applications. These can be flat plate collectors, evacuated tube collector or compound parabolic collectors. In our case, the solar collectors must be vacuum collectors to provide heat at high temperature to the steam generator for the operation of the chiller and with an almost constant efficiency. Evacuated tubes offer the advantage that they work efficiently with high absorber temperatures and with low radiation. The selected type is called ICPC (Integrated Compound Parabolic Concentrator).

ICPC enables to feed the steam generator with a high temperature fluid (set point temperature: around 184˚C (363°F)). The solar receiver can either be operated at a constant flow and a variable outlet temperature (limited or not) or a variable flow rate and a constant outlet temperature. The performance of the receivers is given in Equation 1:

$$\eta = a_0 - a_1 \frac{\Delta T}{SR} - a_2 \frac{(\Delta T)^2}{SR} \qquad (1)$$

with a_0 = 0.735, a_1 = 0.559 and a_2 = 0.004

T: Average of temperature above ambient, °C

SR: Instantaneous solar radiation, W/m^2

To collect more solar thermal energy, the solar receiver must be oriented and tilted in a certain way. There is no incident angle modifier to maximize the solar energy collection. The solar system is firstly designed to provide cooling but will be adapted to provide heating during winter season. The maximum annual solar energy available in Pittsburgh is reached with a 30° collector slope and 0° azimuth: 1,535 kWh/m^2 (487 $kBtu/ft^2$). Based on this optimum and the building's orientation, the chosen azimuth is -15˚ and the chosen slope is 25°. The annual solar energy available is 1,526 kWh/m^2 (484 $kBtu/ft^2$).

3.2 Piping. Piping size is determined based on a recommended maximum velocity in the pipe and flow regime. Piping is in carbon steel. Piping network is insulated with 3.8 cm (1.5 in) of fiberglass insulation.

3.3 Storagen Tank. The storage tank is needed to store heat transfer oil coming from the solar collector in order to increase the efficiency of the system and allow the system to operate when there is no sunshine but heat is available in the storage tank. The main advantage of oil storage tank towards hot water storage tank is that the system does not have to be pressurized. But the oil is expensive and thermal properties of water are better than oil's ones.

The tank is a vertical cylindrical tank with height-to-diameter ratio as 2~3:1 in accord with ASHRAE standards. The capacity of the tank is determined by an empirical rule which suggests that the storage capacity is equal to two hours of peak solar radiation. This storage tank is a one input one output tank, as shown in Figure 2. This configuration enables to improve the thermal stratification in the storage tank. The energy from the solar collection loop can be directly used and stored at the same time. In a two input two output configuration, the collected energy is stored and used after mixing in the tank. The tank is charged and discharged at the same time. Based on different charging and discharging flow rate, the two input two output tank is more a mixing tank than a segregated tank. The main advantage of this other configuration would be the simplification of the control system and the enhancement of the system stability.

Since the daily average ambient temperature is lower than the indoor temperature, the storage tank is kept indoor, thus the energy loss from the storage tank is minimized. The storage tank loss coefficient is set to 0.9 $kJ.hr^{-1}.m^{-2}.K^{-1}$ (0.044 $Btu.hr^{-1}.ft^{-2}.°F^{-1}$) for each nodes of the tank.

3.4Auxiliary Heater.

The gas fired auxiliary heater is used to elevate or maintain the fluid medium temperature at 184°C (363°F) at the inlet of the steam generator. The efficiency of this heater is 0.79.

3.5 Steam generator. The steam generator supplies steam to the absorption chiller. The flow rate of the fluid medium trough the steam generator is variable to keep the fluid medium outlet temperature constant. This fluid temperature is set at 163˚C (325°F), very closed to the steam set point temperature (160°C (320°F)). This "low" temperature enables to take more advantage of the solar collection by decreasing the fluid temperature at the inlet of the solar collectors and then increasing their efficiency.

3.6Absorption chiller. The chiller studied is a two stage, LiBr based, steam fired absorption chiller with an associated cooling tower. A catalog data lookup to predict the performance of this chiller was created with a EES model based on previous testing, and measurements. The rated conditions are:

- cooling capacity: 16.17 kW (55.2 kBtu/hr)
- chilled water temperature: supply 7°C (44.6°F) and return 14˚C (57.2°F)
- chilled water flow rate: 2,044 kg/hr (9 gpm)
- cooling water temperature: supply 30˚C (86°F)
- cooling water flow rate: 4,088 kg/hr (18 gpm)
- steam temperature: 160˚C (320°F)
- steam pressure: 617.8 kPa (89.6 psi)
- COP: 1.18

4. ANALYSIS APPROACH

A base case is defined based on several design assumptions concerning the adequacy of the solar receiver area, the tank capacity, and the chiller capacity to meet the IW_s cooling loads. Then a sensitivity analysis is performed to improve the system design and operation.

4.1Definition of the base case

4.1.1Design parameters. The solar collector area was calculated on a preliminary basis with Pittsburgh's weather data and building loads in Fchart. The solar collector area is set to 40m^2 for the base case.

The maximum solar gain in Pittsburgh by a 40m^2 collector is about 20kW (68.2 kBtu/hr) for one hour solar peak, 40kW (136.5 kBtu/hr) during two hours of peak solar radiation. The design tank volume is given by Equation 2:

$$V = \frac{Q_{solar\_2hr\_peak}}{\rho_{Syl}.Cp_{Syl}.\Delta T_{storage}} \tag{2}$$

$$= \frac{20[kW]*2[hr]*3600[s/hr]}{800[kg/m^3]*1.8815[kJ/kg.°C]*(184-163)[°C]} = 4.55m^3(15\,ft^2)$$

With a height-to-diameter ratio equal to 2, the height of the tank is 2.85 m (9.4 ft) and the diameter is 1.42 m (4.7 ft).

4.1.2Operational configuration. The solar receivers are operated at a variable flow rate and a constant outlet temperature (set point temperature: around 184˚C (363°F)).

4.2 Sensitivity Analysis

Table 3 summarizes the different operational configurations and design parameters.

Tab 3. Sensitivity parameters

OPERATIONAL CONFIGURATIONS	
Configuration A	change the flow rate to keep the solar collector outlet temperature constant at 184°C (363°F)
Configuration B	keep the flow rate constant from 1,000 to 3,000 kg/hr (5.5- 16.5 gpm) without high cut limit temperature
Configuration C	keep the flow rate constant from 1,000 to 3,000 kg/hr (5.5- 16.5 gpm) with high cut limit temperature from 200 to 220°C (392-428°F)
DESIGN PARAMETERS	
Collector area	20 m^2 - 60 m^2 (215 ft^2- 646 ft^2)
Tank volume	2hr– 1.5 hr – 1 hr – 0.5 hr – 0.25hr solar peak

4. 3Analysis indicators

Optimum design and operation of the solar system are determined based on energy and economic indicators.

Energy indicators. The performance of the solar system is characterized by:

- the solar fraction. The fraction of load covered by the solar system is given by Equation 3:

$$Solarfraction = \frac{Qload - Qaux}{Qload} \qquad (3)$$

with Q_{aux} Auxiliary energy and Q_{load} Cooling energy The power consumption of circulating pump and controller is excluded.

- the saved energy per square meter of solar array. This saved energy is the solar energy used divided by the solar receiver area.

4.4Economic Indicators.

The economic indicator is the life cycle cost. The assumptions for the investment cost are shown in Table 4. The cost of the storage tank, insulations, pumps and pipes are not included in the system costs because they play a negligible part in the overall system price. The cost of the storage tank is relatively small due to the low pressure of the oil. The main investment costs are the solar collectors and the fluid medium.

Tab 4. Investment cost of the system's main components

Component	Cost ($)
Solar collector	200 $/$m^2$ (18.6 $/$ft^2$)
Storage tank, oil, insulation	The main cost is the oil 8,214 $/$m^3$ (31.1 $/gal)
Other equipment (pumps, pipe)	Not included in this system costs

Reducing the storage capacity enables to decrease the amount of oil so reduce the cost of the storage as shown in Table 5.

Tab 5: Cost of the storage for different capacities

Volume of the tank		Price of oil ($)
2hr	4.55 m^3 (1,200 gal)	37,374
1.5hr	3.41 m^3 (900 gal)	28,031
1hr	2.28 m^3 (600 gal)	18,687
0.5hr	1.14 m^3 (300 gal)	9,344
0.25 hr	0.57 m^3 (150 gal)	4,672

Further research of the cost of piping and pumping would increase the precision of the life cycle cost. The life time of the system is assumed to be 20 years. The energy savings are calculated based on tables of projected nominal (i.e., including inflation) fuel price indices for electricity in the commercial sector for each of the years from 2005 through 2025. These price indices are based on the DOE energy price projections used to calculate the FEMP UPV* factors for energy costs. The indices include the effect of four alternatives, hypothetical rates of general price inflation IR: 2, 3, 4, and 5 %. Uses of the indices produce price estimates in current dollars, inclusive of general price inflation. The real rate considered (excluding general price inflation) is 3 % (DOE discount rate for 2005). Table 6 enables to calculate more precisely the energy cost savings based on discount and inflation rates.

Tab. 6: Estimation of the cost of electricity for a commercial HVAC system in Pennsylvania from 2005 to 2025, given the energy cost in 2005 prices and different inflation rates

2005 prices (Carnegie Mellon University)	Electricity = 0.07 $/kWh (20.5 $/MBtu)
Projected fuel prices with assumed general price inflation rates of 2, 3, 4 and 5% for a 20 year life cycle	Electricity IR 2% : 0.0805 $/kWh (23.6 $/MBtu) IR 3% : 0.0903 $/kWh (26.5 $/MBtu) IR 4% : 0.1015 $/kWh (29.8 $/MBtu) IR 5% : 0.1141 $/kWh (33.5 $/MBtu)

The investment cost and the cost of the electric energy saved during the system life time are used to calculate the life cycle cost (LCC), see Equation 4:

LCC ($) = Life time (years) * annual energy savings (kWh/year)* energy cost with inflation ($/kWh) – investment cost ($) (4)

Life cycle period = 20 years

Annual energy savings = solar energy collected and used

Energy cost (electricity) = see Table 6

Investment cost = solar array area (m^2) * 200 $\$/m^2$ + tank volume (m^3) * 8214 $\$/m^3$= solar array area ($ft^2$) * 18.6 $\$/ft^2$ + tank volume (gal) * 31.1 $/gal

5 .RESULTS OF THE BASE CASE

The results of the base case simulation are shown in Figure 3. The base case system can provide 75 % of the thermal energy required to generate the cooling to meet the building loads. 27 % of the solar energy used to meet the space cooling requirements are provided by the storage tank.

The heat losses in the system represent 31 % of the solar energy collected. This high percentage is due to:

- the high temperature of the oil in the pipe and the tank (184 ˚C),
- the high exchange area (capacity of the storage tank V = 4.55 m^3)
- and that there are no loads during the night. The energy stored in the pipe is lost during the night.

The collector efficiency is 42% and the energy saving per square meter of solar collector is 255 kWh/m^2.

The life cycle cost of this system is from -27,844 to -20,527 $ for energy price inflation rates from 2% to 5%. Even if the solar fraction of the base case is high, the investment costs are too high to get a positive life cycle cost.

6 SYSTEM OPTIMIZATION

A sensitivity analysis on design and operation parameters is performed to improve the base case. TRNSYS simulations are performed for three different operational configurations. Design parameters (collector area and storage tank volume) are varied to determine optimal settings for each operational configuration. Results are compared to define the optimum design and operation for the IW_s solar driven cooling system. Table 7 summarizes the studied operational configurations.

Tab 7. Summary of the different configurations

Configuration A : Change the solar collector flow rate to keep the solar collector outlet temperature constant
Configuration B: Keep the solar collector flow rate constant without high limit temperature
Configuration C: Keep the solar collector flow rate constant with high limit temperature

6.1 Configuration A analysis

In the configuration A, the solar collector is operated at a variable flow rate to maintain a constant fluid temperature 184°C (363°F) at its output.

6.1.1 Energy analysis. Figure 4 shows the variation of the monthly solar fraction for different solar collector areas and the design storage tank volume (2hr solar peak: 4.55 m^3 (1,200 gal)) during the cooling period. Small collector areas are not able to meet large summer cooling loads. Collector efficiencies decrease as collector area increases. The larger systems have poorer integrated efficiencies as they are oversized during spring and fall.

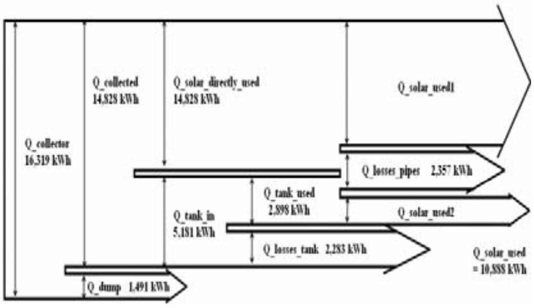


Fig 3. Base case results

Thermal energy	kWh
Space cooling demand	15,176
Total heat demand for steam generation	13,603
Total solar energy collected	14,828
Total auxiliary energy	3,415
Total heat loss in the system	4,640
Total solar energy used	10,188
Performance parameters	
Solar fraction	0.75
Saved energy per collector area	255 kWh/m^2
Collector efficiency	42%

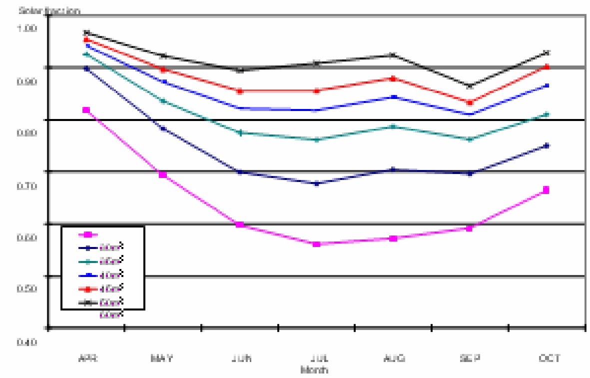


Fig. 4. Monthly solar fraction for different solar collector areas with design storage tank size (configuration A)

Figure 5 shows the variation of the solar fraction for different solar collector areas and storage tank volumes during the cooling period. For a given tank size, increasing the solar collector area leads to increase the solar fraction: more solar energy is collected and less auxiliary heat is required to meet the cooling loads. The optimum solar collector area can only be determined thanks to economic analysis.

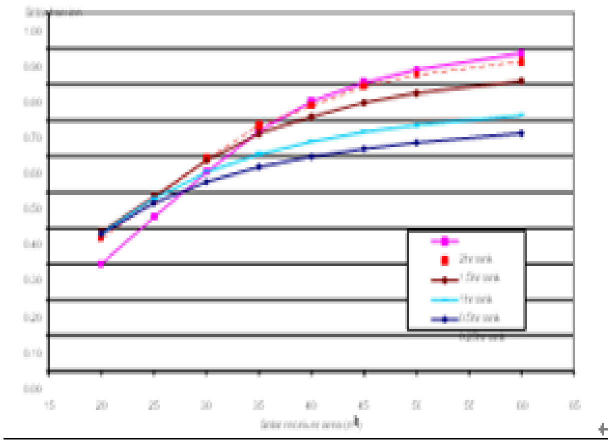


Fig 5. Evolution of the solar fraction for different solar collector areas and storage tank volumes (configuration A)

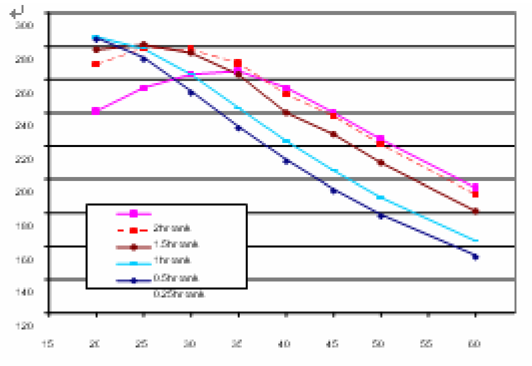


Fig 6. Evolution of the energy savings for different solar collector areas and storage tank volumes (configuration A)

Figure 6 shows the energy savings per solar array area (in kWh/m^2) for different solar collector areas and storage tank volumes during the cooling period. Figure 5 and 6 show the existence of an optimum tank volume for a given solar collector area. Table 8 summarizes the optimum tank volume for a given solar collector area and the system performance (solar fraction, energy savings). A proper size of the storage tank is essential. For large solar array, more solar energy is collected and has to be stored in a big tank. For small solar area, the solar energy collected is small and has to be stored in a small tank to decrease exchange area and the heat loss in the tank.6.1.2 Economic analysis. The study shows that, to ensure the reliability of the system design, a proper size of the storage tank is essential, and an optimum ratio of tank over collector area exists.

Tab 8. Optimum design parameters to improve energy performance (configuration A)

Solar receiver area		Optimum tank volume	Solar ratio	Energy Savings	
m^2	ft^2	solar peak hour	-	kWh/m^2	$kBtu/ft^2$
20	215	0.5 (0.25)	0.43	284	90
25	269	0.5 (0.25)	0.50	281	89
30	323	1.5 (1)	0.59	278	88
35	431	1.5	0.66	270	86
40	484	2	0.75	255	81
45	538	2	0.83	240	76
50	592	2	0.86	224	71
60	*646*	2	0.89	195	*62*

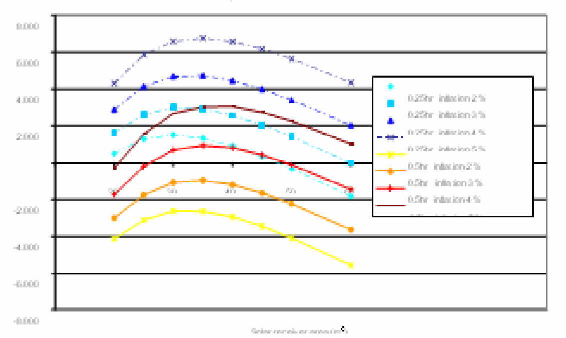


Fig. 7. Evolution of life cycle cost for different solar collector areas, storage tank volumes and inflation rates (configuration A)

To determine the optimum collector area and the storage tank size, one economic study must be performed to ensure the profitability of the system. The study of the inflation rate enables to realize that some system can be profitable depending on the future cost of the energy. Due to the use of expensive oil, the storage capacity has to be decreased to reduce the cost of storage. Only 0.5hr and 0.25hr can have positive life cost cycle. Figure 7 shows the evolution of life cycle cost for different solar collector areas, 0.5hr and 0.25hr storage tank. Optimum numerical values are shown in Table 9.The life cycle cost is positive for a system with a 0,6m^3 storage tank whatever the energy inflation rate is. Having a 0.25hr storage tank is more cost efficient than a 0.5hr one for the same inflation rate. The extra energy savings achieved by a system with 0.5hr tank are not enough to cover the investment cost of the extra fluid medium.

For a given inflation rate, the proper design of the solar cooling system for configuration A is:

- solar area : 30-35 m^2
- storage tank volume : 0.25hr (0.6m^3)

Tab 9. Economical design parameters (solar collector area and storage tank volume) for different inflation rates (configuration A)

Storage tank volume	Inflation rate	Optimum area (m^2)	LCC ($)	Solar ratio
0.25 hr solar peak 0.6 m^3 (150 gal)	2%	30	1,498	0.53
	3%	30	2,979	0.53
	4%	35	4,717	0.57
	5%	35	6,751	0.57
0.5hr solar peak 1,2 m^3 (300 gal)	2%	Negative life cost cycle		
	3%			
	4%	35	921	0.61
	5%	40	3,035	0.64

For a 2% inflation rate, the solar thermal system can supply 53 % of the heat required to drive the chiller. The system investment cost is around 10,700 $ for a 30 m^2 solar array and the life time energy savings are 12,198 $. So the life cycle cost is 1,498 $. Solar collector price represents 56% of the system investment cost and storage tank price 44%. The energy savings is 7,559 kWh (only 10% is coming from the storage tank, 733 kWh). The storage tank losses are 45 % of the total energy entering in the tank (608 kWh on 1342 kWh). The piping losses are 30% of the solar energy collected by the solar receiver. The storage tank cost is greater than the percentage of the total energy used coming from the tank. The storage tank with oil seems to be too expensive.

6.2 Configuration B analysis.

Energy and economic analyses. In the configuration B, the solar collector is operated at a constant flow rate without a high limit cut temperature at its output. The effect of the solar pump flow rate on the solar fraction has been studied by simulating the solar system with different constant flow rates from 1,000 to 3,000 kg/hr (5.5- 16.5 gpm).

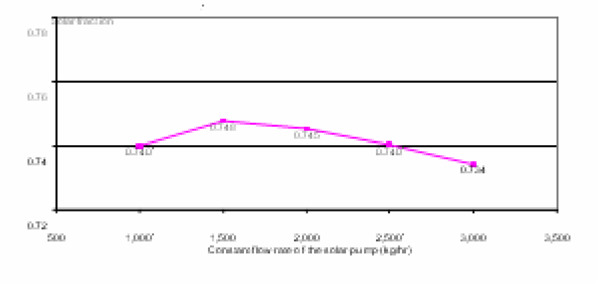


Fig. 8. Effect of the solar pump constant flow rate on the solar fraction (configuration B)

Tab. 10. Optimum design parameters to improve energy performance (configuration B)

Solar receiver area		Optimum tank volume	Solar ratio	Energy Savings	
m^2	*ft^2*	solar peak hour	-	kWh/m^2	*kBtu/ft^2*
20	*215*	0.25	0.35	263	*83*
25	*269*	0.5	0.47	268	*85*
30	*323*	1 (0.5)	0.57	268	*85*
35	*431*	1	0.67	266	*84*
40	*484*	1.5	0.77	260	*82*
45	*538*	1.5	0.85	250	*79*
50	*592*	2	0.89	239	*76*
60	*646*	2	0.98	211	*67*

Increasing the flow rate of the solar pump leads to lower fluid temperature at the outlet of the solar collector, lower steam generation and thus lower chiller performance. Whereas decreasing solar pump flow rate leads to higher driving temperature, higher chiller performance and better tank thermal stratification. However the flow rate should be enough to provide the energy required to meet the cooling loads and the temperature should not be too high to limit the heat losses in the system. Figure 8 shows that the maximum solar fraction is reached with a constant flow rate of 1,500 kg/hr (8.25 gpm). This optimum flow rate is used to generate the following results. As for previous simulations, the results show the existence of an optimum tank volume for a given solar collector area. Table 10 summarizes the optimum volume tank for each solar collector area with the energy performance and savings achieved.

Fig 9 shows the evolution of life cycle cost for different solar collector areas, and storage tank volumes (0.5hr and 0.25hr). Table 11 shows the economical design parameter to improve the energy performance and positive life cycle cost.

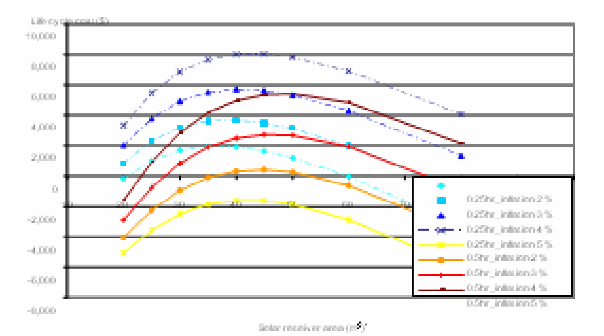


Fig 9. Evolution of life cycle cost for different solar collector areas, storage tank volumes and inflation rates (configuration B)

The life cycle cost is positive for a system with a 0,6m^3 storage tank whatever the energy inflation rate is. For a given inflation rate, the proper design of the solar cooling system for configuration B is:

- solar area : 35-40 m^2
- storage tank volume : 0.25hr (0.6m^3)

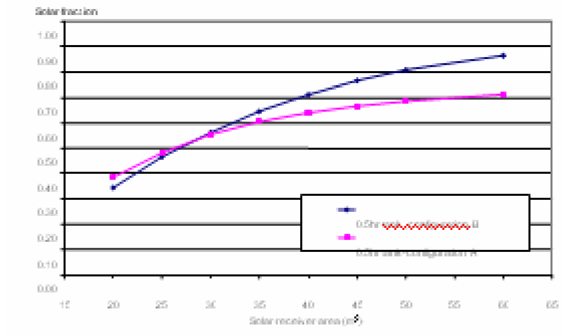
Operating a solar cooling system with 1.2 m^3 hot storage tank instead of 0.6 m^3 leads to a 20% increase of the solar fraction. But the investment cost due to the enlargement of the storage tank is still higher than the additional annual energy savings.

Tab 11. Economical design parameters (solar collector area and storage tank volume) for different inflation rates (configuration B)

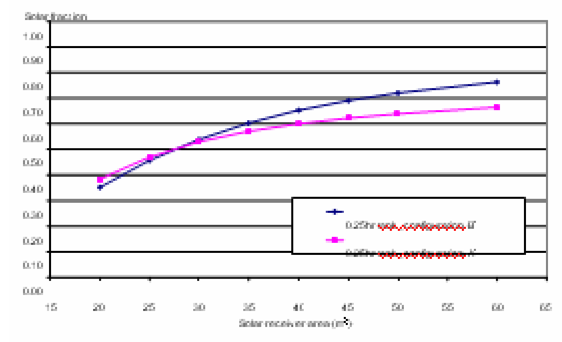
Storage tank volume	Inflation rate	Optimum area (m^2)	LCC ($)	Solar ratio
0.25 hr solar peak 0.6 m^3 (150 gal)	2%	35	1,962	0.60
	3%	40	3,690	0.65
	4%	40	5,720	0.65
	5%	40	8,003	0.65
0.5hr solar peak 1,2 m^3 (300 gal)	2%	Negative life cycle cost		
	3%	45	404	0.77
	4%	45	2,729	0.77
	5%	50	5,400	0.81

7.3 Comparison configurations A and B. Figures 10a and 10b enables to compare the system performance for configurations A and B for a given tank volume and solar collector area.

The energy performance achieved in configuration B is higher than in configuration A when the solar collector area is greater than 28 m^2. Based on the previous results, economical solar collector area is greater than 28m^2 both in configuration A and B. The configuration B is then more efficient and more profitable than the configuration A. Keeping the outlet temperature of the solar collector constant by varying the flow rate (configuration A) prevents the system from overheating. Keeping the flow rate constant (configuration B) enables to simplify the operation and control of the solar receiver. To increase the heat transfer and optimize the energy performance of the system, the constant oil flow rate has to be low (1,500 kg/hr). This system operation leads to very high temperature generation in the system. Figure 11 shows the maximum oil temperature in the system for a given flow rate (1,500 kg/hr) and different design parameters.



(a)



(b)

Fig 10..Effect of the solar receiver area and the type of the tank on the solar ratio for each tank volume

(a) Config.A: variable flow rate to maintain 184˚C

(b) Config. B: constant flow rate solar pump (1,500 kg/hr)

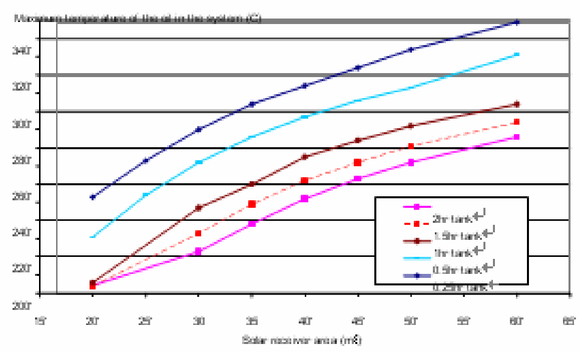


Fig. 11. Maximum temperature in the system for different design parameters (configuration B)

The temperature increase is acceptable as long as the system devices (piping network, pump, the storage tank, the solar collector) can handle this maximum temperature. High temperature can be handled by an oil thermal fluid but not with water

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(too high pressure). To prevent overheating and device damages, a high limit cut temperature can be set (configuration C).

6.3 Configuration C Analysis

In the configuration C, the solar collector is operated at a constant flow rate (1,500 kg/hr) with a high limit cut temperature from 200 to 220°C (392-428°F) at its output.

Figure 12 shows the effect of the different high cut limit temperature on the solar thermal system performance. Using a high limit cut temperature enables to decrease the risk of overheating but it also decreases a lot the solar thermal system performance especially for system with large solar collector array and small storage tank. Those systems generate too high temperature as shown previously in Figure 11.

Figure 13 shows the evolution of life cycle cost for different solar collector areas, and a given storage tank volume (0.25hr). Tab. 12 shows the economical design parameter to improve the energy performance and positive life cycle cost.

For a 3% inflation rate, the solar collectors can supply 38 % of the heat required to drive the chiller. The life time savings are 519 $. The system investment cost is (for 25 m^2) around 9,670 $ (52 % solar collector and 48 % storage tank).The annual energy savings are 5,643 kWh. The storage tank losses are 9 % of the total energy entering in the tank (622 kWh on 7248 kWh). The pipe losses are 1,580

7. CONCLUSIONS

The TRNSYS cooling system model has provided performance information that has proved useful in optimizing design and operating conditions and in suggesting alternate configurations and equipment selections for this solar thermal driven cooling system.

The building simulation indicates that the 16.17 kW (55.2 kBtu/hr) absorption chiller is adequate to meet the IW south cooling requirements (sensible loads). The energy supply system simulations indicate that:

- the base case design wasn't appropriate due to a high investment cost; the storage tank was too big.
- estimated heat losses from the system can reach about 20-30% of the total heat collected by the solar receiver. to be cost effective,

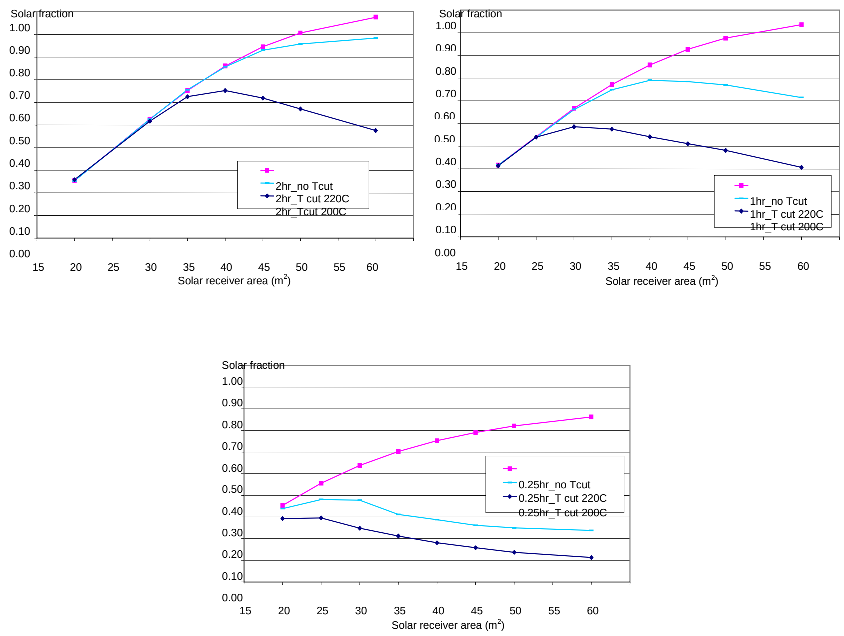

Fig. 12. Effect of the high cut limit temperature on the solar fraction for different design parameters(without Tcut : Configuration B / with Tcut: Configuration C)

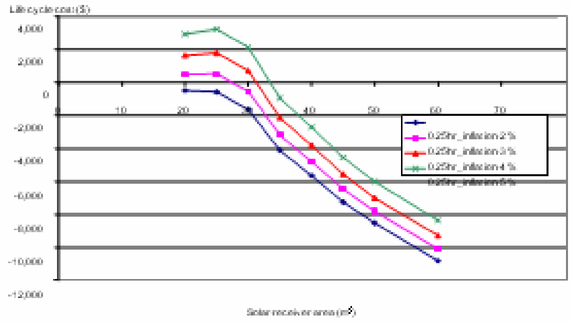

Fig. 13. Evolution of life cycle cost for different solar collector areas and inflation rates (configuration C)

kWh (20% of the solar energy collected by the solar receiver).

Tab. 12. Economical design parameters (solar collector area and storage tank volume) for different inflation rates (configuration C)

Storage tank volume	Inflation rate	Optimum area (m^2)	LCC ($)	Solar ratio
0.25 hr solar peak 0.6 m^3 (150 gal)	2%	Negative life cycle cost		
	3%	25	519	0.38
	4%	25	1,783	0.38
	5%	25	3205	0.38

the solar collector area has to be, depending on the operation strategy, between 30 and 40 m^2. The storage tank volume has to be small (0.6 m^3) due to the high cost of the fluid medium.

They also indicate that the solar energy collected can reach:

- for a variable solar pump flow rate and constant operating temperature's control strategy (configuration A): from 53 % to 57 % of the heat required to drive the chiller
- for a constant solar pump flow rate and variable operating temperature's control strategy: from 60% to 65 % for high operating temperature system (around 300°C) (configuration B) and 38% for a lower operating temperature system (around 220°C) (configuration C) of the heat required to drive the chiller.

So for high temperature resistant devices (300°C), operational configuration B is more appropriate than configuration A; whereas for lower heat-resistant devices, operational configuration A is more appropriate than configuration C.

Further researches are currently performed to improve the performance of this solar thermal system. The life cycle cost of the system can be increased by:

- operating the system all year long for space heating and cooling without critical additional investment cost
- decreasing the cooling loads thanks to passive cooling system (natural ventilation)
- decreasing the heat losses in the system by increasing insulation thickness and privileging cold storage to hot storage
- collecting more thermal energy thanks to solar tracking devices and water as heat transfer fluid

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